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DUAL REGENERATIVE COOLING CIRCUITS FOR LIQUID ROCKET ENGINES (Preprint)

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Effectiveness of dual cooling to lower the maximum wall temperature of regeneratively cooled engines is the focus of this study. Two engines, the SSME and a RP1-LOX engine, are retrofitted with dual-circuits. It is shown that the maximum wall temperatures for both engines are substantially reduced while also lowering coolant pumping power. It is also shown that with RP1 as the coolant, the likelihood of coking is reduced with use of dual-circuits.

I.   Introduction

For high-pressure liquid rocket engines (LRE's), hot-gas in the throat area may reach temperatures as high as 7000 ºR. Therefore, it is essential to cool the engine ensuring that the wall material withstands the high temperatures. In addition, using the fuel/oxidizer as the coolant increases the enthalpy prior to combustion, resulting in a more efficient combustion. Single Circuit Regenerative cooling is a widely used method to reduce the wall temperatures and increase coolant enthalpy for high-pressure LRE's [1].

Given this regenerative cooling method, the coolant is either fuel or oxidizer and the flow path of the fluid is shown in Figure 1. The coolant first enters cooling passages at the nozzle exit and travels through the passages to exit at the nozzle entrance. This method serves two purposes: 1) keeps the engine walls cool and, 2) increases coolant enthalpy. *In some engines, such as the SSME, the coolant (LH_2) coming out of cooling channels is also used to run turbo-pumps.*

Presently, nearly all regeneratively cooled LRE's have only one cooling circuit (Figure 1.). When the coolant reaches the throat area (i.e., largest heat flux region) from the nozzle exit, it is heated to a high temperature, lowering its cooling capability. This cooling arrangement, known as Counter-Flow Cooling, works well due to the following reasons:

- The coolant being fuel or oxidizer is used for combustion.
- Having the exit manifold close to the injector simplifies the coolant manifold design.
- The distance that the coolant travels in the diverging section of the engine is shorter than that of combined chamber and converging sections. Hence, it absorbs less thermal energy by the time it reaches the throat area.

For a dual-circuit regeneratively cooled LRE, the coolant enters the cooling channels at the high heat flux region (throat area). The coolant splits into two separate cooling circuits; where one circuit travels downstream and the other travels upstream of the throat (Figure 2). Dual-circuit cooling offers the following advantages over the single-circuit method:

- The coolant temperature is the lowest at the highest heat flux region, providing maximum cooling.
- The coolant heat transfer coefficient is large at the channel entrance.
- It is possible to use a fuel as the coolant in one circuit, and an oxidizer in the other.
- The downstream circuit can be used as a dump cooling circuit.

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The sole disadvantage between the single-circuit and dual-circuit cooling system is the manufacturing cost associated with extra manifolds for the dual-circuit.

This paper demonstrates the effectiveness of co-current flow and dual-circuit systems. The combined TDK and RTE codes, described in References [2], [3] and [4], are modified to include the capability of a dual-circuit analysis. The modified TDK-RTE model is used to examine the thermal characteristics of a dual-circuit design for the SSME and RP1-LOX engines.
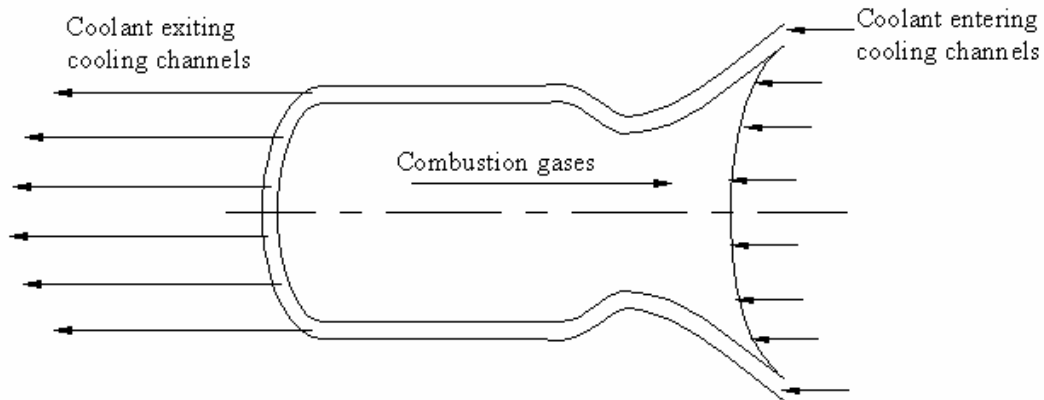
Figure 1. Schematics of a single-circuit counter-current regenerative cooling commonly used in rocket engines.
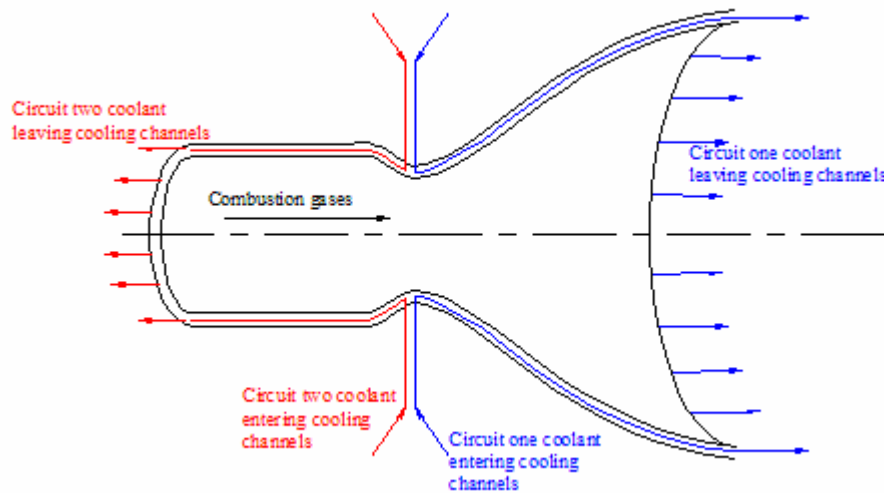
Figure 2. Schematics of a dual-circuit regenerative cooling.

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II. Dual-Circuit Regenerative Cooling

A) Dual-circuit cooling with the same coolant

To demonstrate the effectiveness of dual-circuit cooling, the SSME is retrofitted with several dual-circuit designs concepts. The first design evaluates a single coolant, liquid hydrogen, which enters the cooling channels at the station with the largest heat flux, i.e., at x = -0.8". Then the coolant splits into two different circuits, one flowing downstream and the other upstream of the throat. For the SSME using the original single-circuit coolant design, the total flow rate is 29.06 lb/s. For the SSME using the dual-circuit design, 6 lb/s of the liquid hydrogen flows through the downstream cooling circuit and 23.06 lb/s through the upstream circuit. Note that the larger coolant flow rate is for the upstream cooling circuit, since the overall heat transfer from the hot-gases is substantially larger in the engine's chamber and converging sections than that of the diverging region.

Since the local flow rates in the cooling passages of the dual-circuit design are lower than that of the original single-circuit design, the cooling channel dimensions are reengineered to accommodate lower flow rates. Figure 3 shows coolant passage dimensions for both the single and dual circuit designs. The cooling passage width and height for the single-circuit design are left the same as the original SSME engine. For the reference case, the cooling passage width for the dual-circuit design is kept the same as the original SSME design. However, for the dual-circuit case, the height of the passage is redesigned such that the height at the entrance is 0.2" reducing to approximately 0.06" and 0.07" for the upstream circuit and approximately to 0.025" for the downstream circuit. The redesign of the cooling channel is accomplished through an iterative procedure of RTE runs by varying channel heights and flow rates. The main objectives of the iterations are to keep the wall temperature as low as possible while maintaining the coolant pressure drop and Mach numbers within a design range.
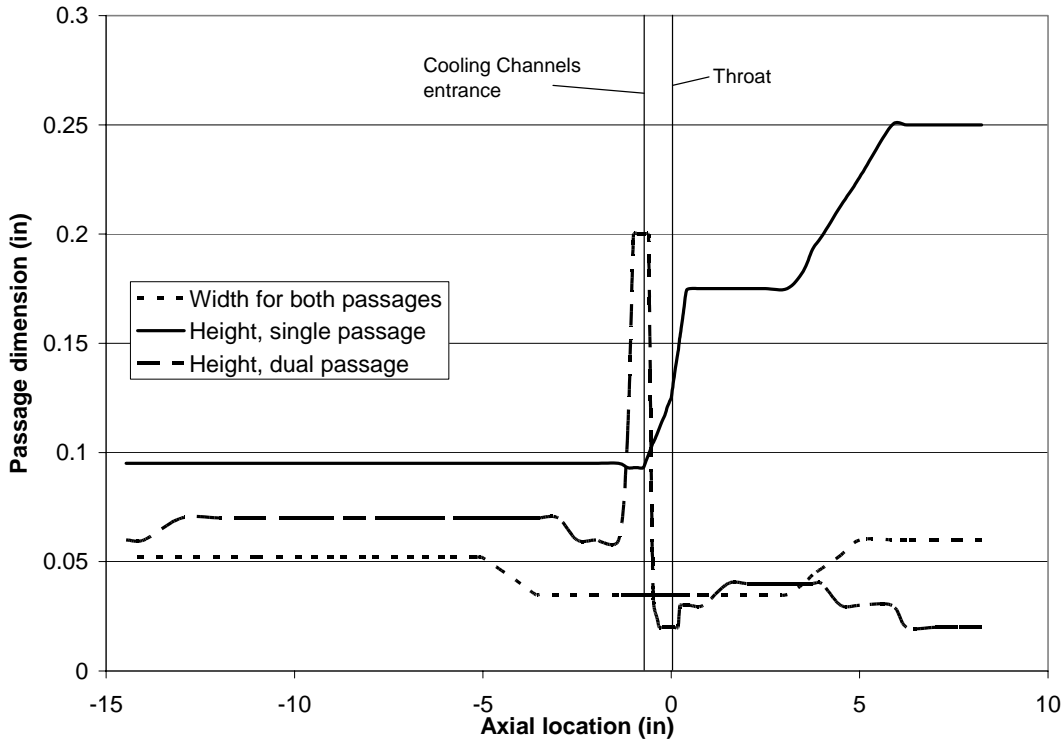

Figure 3. Cooling passage width and height of the SSME for the original single-circuit and dual-circuit arrangements.

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Figure 4 shows the resulting maximum wall temperature distributions along the axial direction computed by the TDK-RTE code, given both the original single-circuit and the new dual-circuit designs. As shown in Figure 4, the maximum wall temperature for the SSME in the region of the throat for the dual-circuit cooling passage is about 120 °R less than that for the original single-circuit design. The results of Figure 4 reveal that the temperature difference between the maximum and minimum wall temperature for the dual-circuit design is smaller than that of single-circuit. The dual-circuit case shows that the lowest hot-gas-side wall temperature is about 920 °R and the highest temperature is 1250 °R, resulting in a 330 °R temperature variation in the axial direction. However, for the single-circuit cooling channel, the lowest hot-gas-side wall temperature is 600 °R and the highest temperature is 1350 °R, resulting in a 750 °R temperature variation in the axial direction.
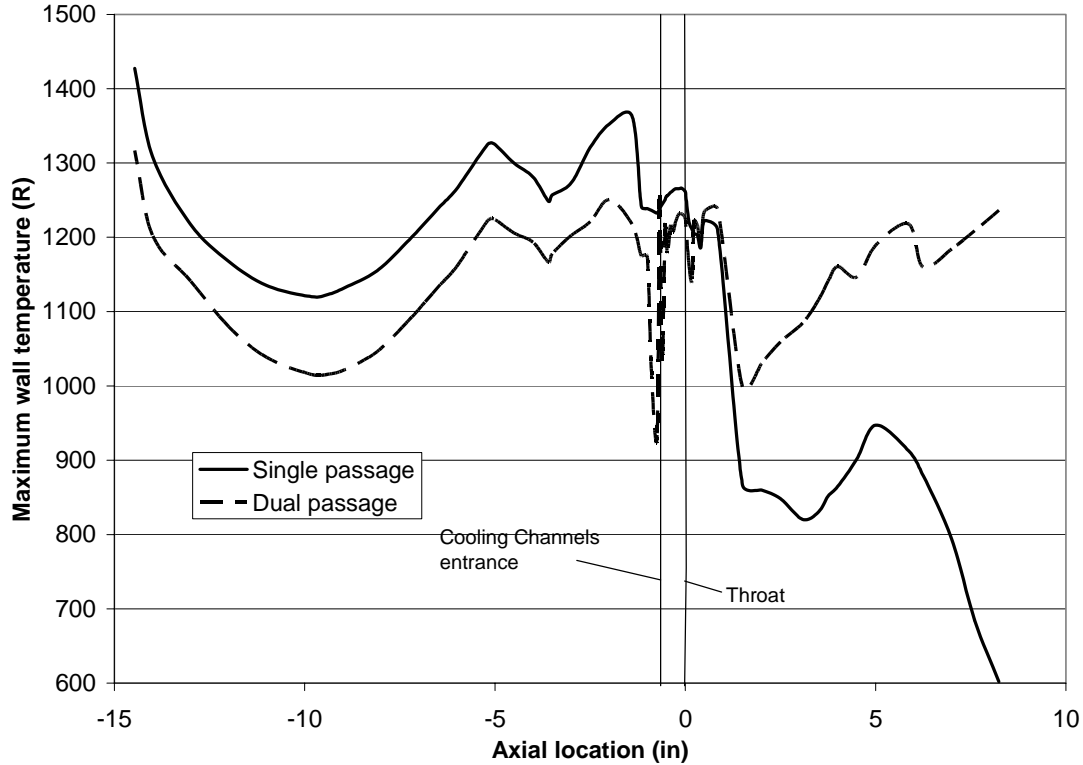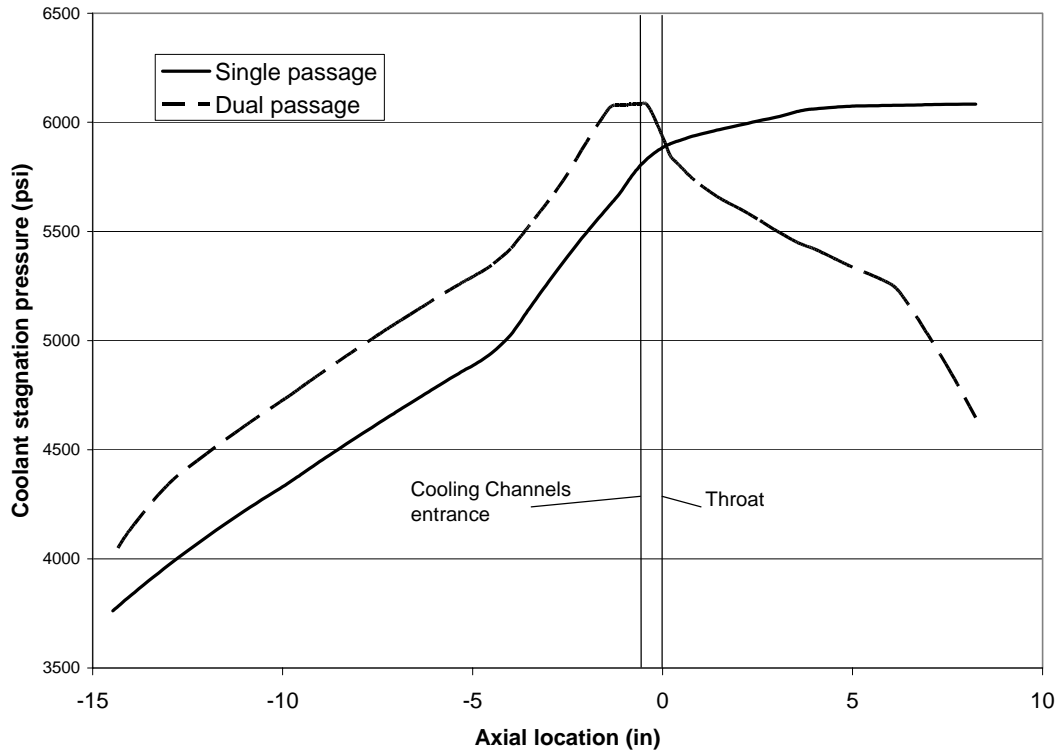

Figure 4. Wall maximum temperature distributions for the SSME original single-circuit and dual-circuit cooling designs.

The distributions of wall heat fluxes along the axial direction for both designs are shown in Figure 5. This figure shows that the maximum wall heat flux for the dual-circuit design is more than that of a single-circuit. This is due to the lower wall temperature for the dual-circuit design at the throat region.

Figure 6 shows the coolant pressure distribution for both the original single-circuit and dual-circuit designs. As shown in this figure, the pressure drop for the single–circuit design is more than 2000 psi, while for the dual-circuit both cooling passages measure a substantially lower pressure drop.

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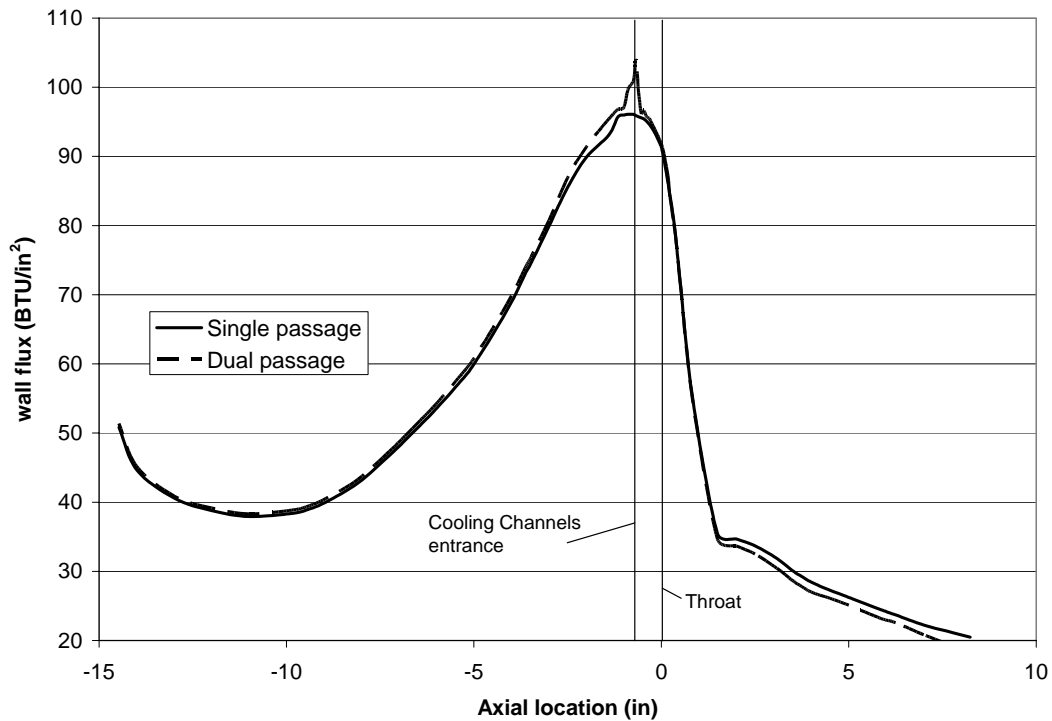
Figure 5. Wall heat flux distributions for the SSME original single-circuit and dual-circuit cooling designs.

Figure 6. Stagnation pressure distributions for the SSME single-circuit and dual-circuit cooling designs.

The coolant temperature distributions along the axial direction are shown in Figure 7. This figure shows the coolant exit temperature for the single-circuit cooling channel design as 566 ºR, while the dual-circuit design exit temperatures are 803 ºR and 501 ºR. Using Equation 1 below, the mixture temperature of the dual-circuit design yields a temperature of 599 ºR; resulting in an exit temperature close to that of the single-circuit design.

$$T_{mixture} = \frac{(c_p \dot{m} T)_{downstream} + (c_p \dot{m} T)_{upstream}}{\overline{c}_p (\dot{m}_{downstream} + \dot{m}_{upstream})} \qquad (1)$$
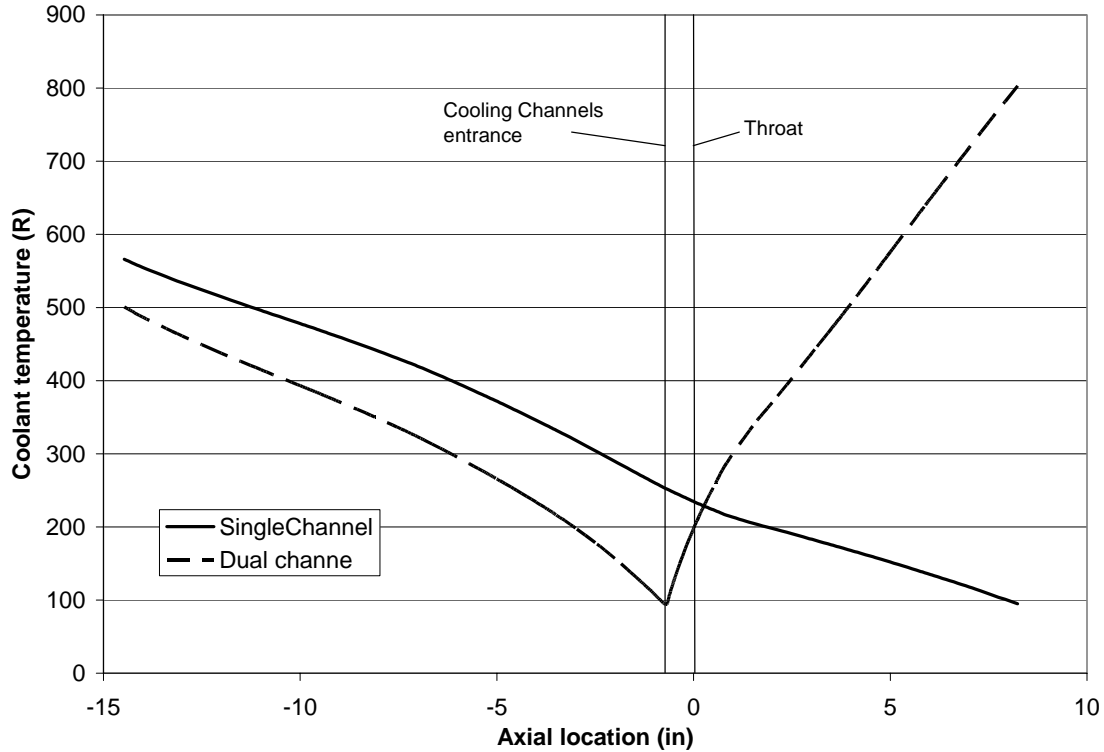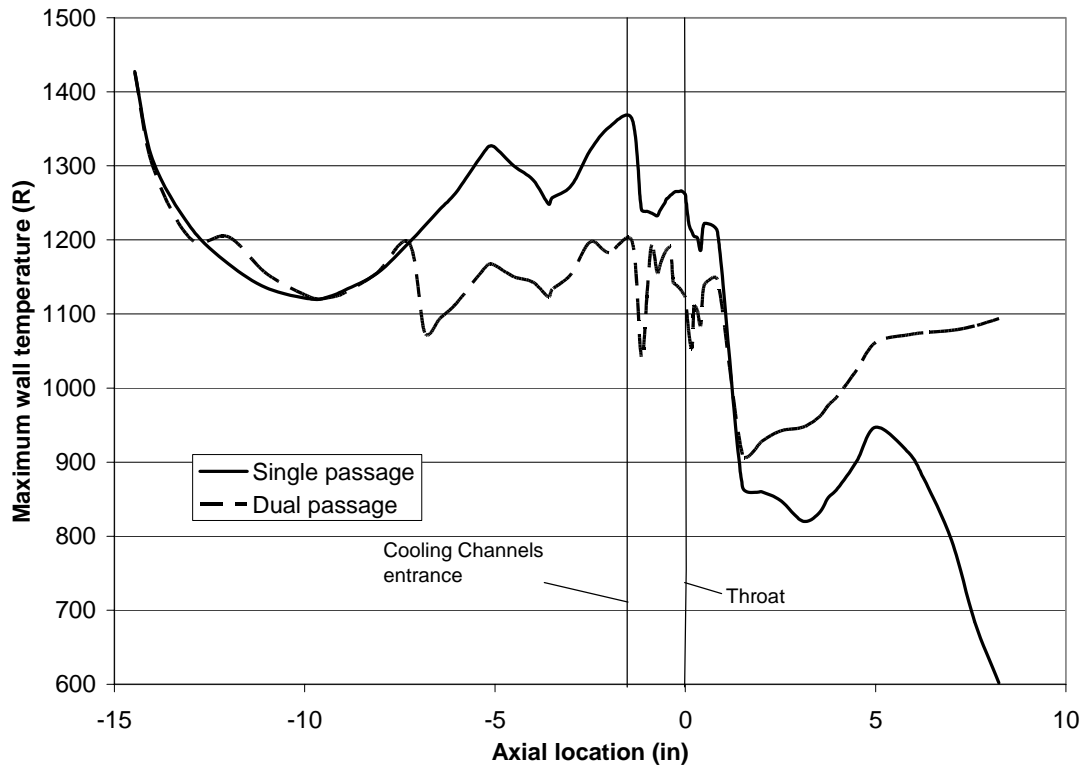

Figure 7. Coolant temperature distribution for single and dual circuit channel designs.

In an attempt to lower the maximum wall temperature, a second dual-circuit design places the entrance of the cooling channels at the location where the highest temperature is calculated for a single-circuit design. For the conventional single-circuit design for the SSME, the maximum temperature of 1363 ºR occurs at axial location x = -1.4". The maximum wall temperature distributions along the axial direction are shown in Figure 8, and indicate that the wall temperature can be reduced by 163 ºR. Furthermore, a reduction in maximum wall temperature can be accomplished by using a lower coolant pressure drop than that of single-circuit design, as shown in Figure 9.

The second dual-circuit design included the new cooling channel dimensions (as described previously) and is shown in Figure 10. The channel widths for both cooling circuits remain the same size as that of the single-circuit design. The cooling channel heights are varied to keep the wall temperature and coolant pressure drop within the design range. Note that the channels heights are large at the entrance of the passage to accommodate the entrance manifold.

As expected, the wall heat flux in the region of the throat for the second dual-circuit design is slightly higher than that of single-circuit, as shown in Figure 11. This is because the maximum wall temperature of the dual-circuit design is more than 150 ºR lower than the single-circuit, as calculated by Equation (1). Finally, the coolant temperatures at the cooling channels exit for the dual-circuit design are close to that of

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the single-circuit exit temperature, as shown in Figure 12. Again, as compared to the single-circuit design, the temperatures of the dual-circuit system are slightly higher downstream of the throat, and lower upstream of the throat.

Figure 8. Maximum wall temperature distributions for the SSME's original single-circuit and dual-circuit cooling designs, with the cooling channel entrances placed at the single-circuit highest temperature location.

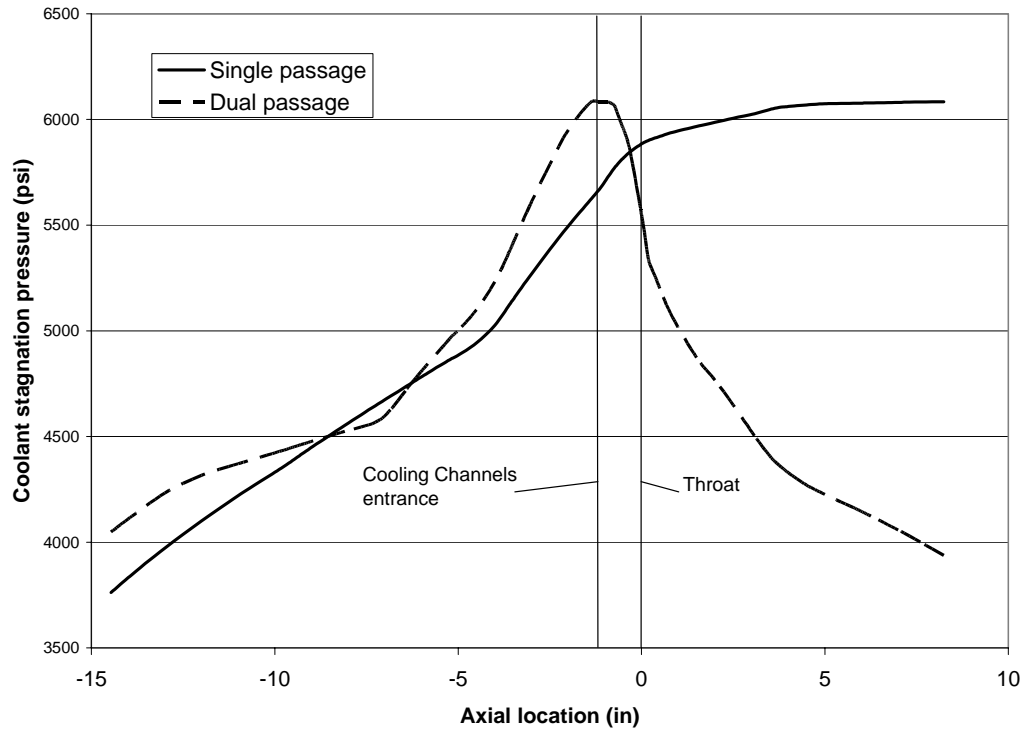American Institute of Aeronautics and Astronautics

Figure 9. Coolant pressure distribution for the SSME dual-circuit cooling channel (channel entrances are placed at the maximum temperature location).
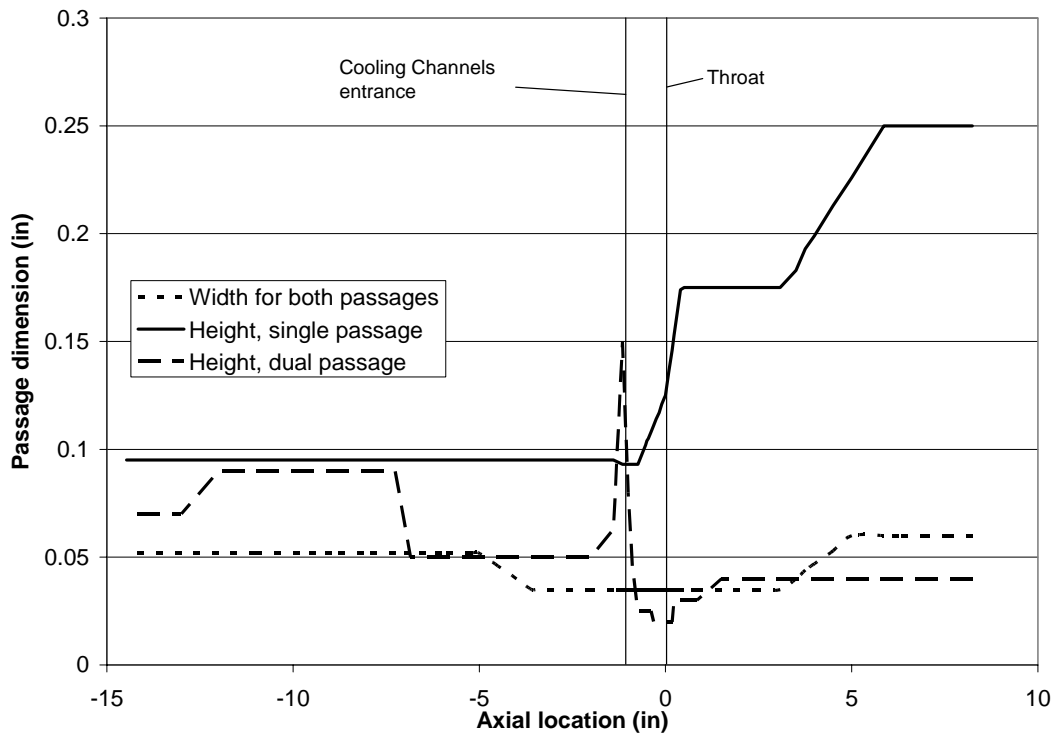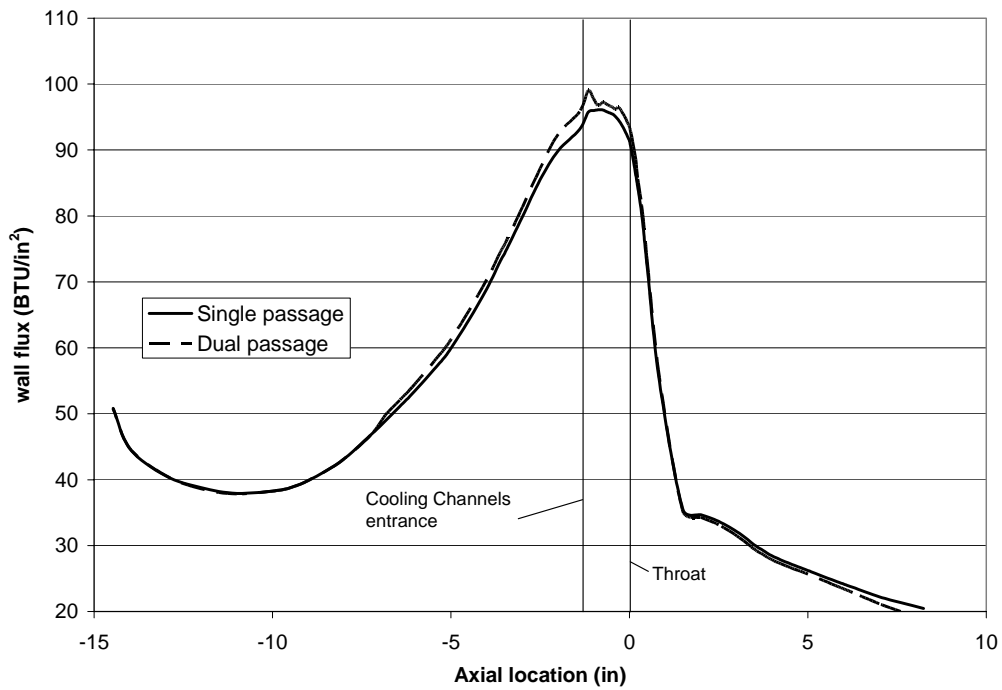

Figure 10. Cooling passage width and height of the SSME for the original single-circuit and dual-circuit arrangements (cooling channel entrances are placed at the maximum temperature point).

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Figure 11. Wall heat flux distributions for the SSME's original single passage cooling passage and dual cooling designs (cooling channel entrances are placed at the maximum temperature point).
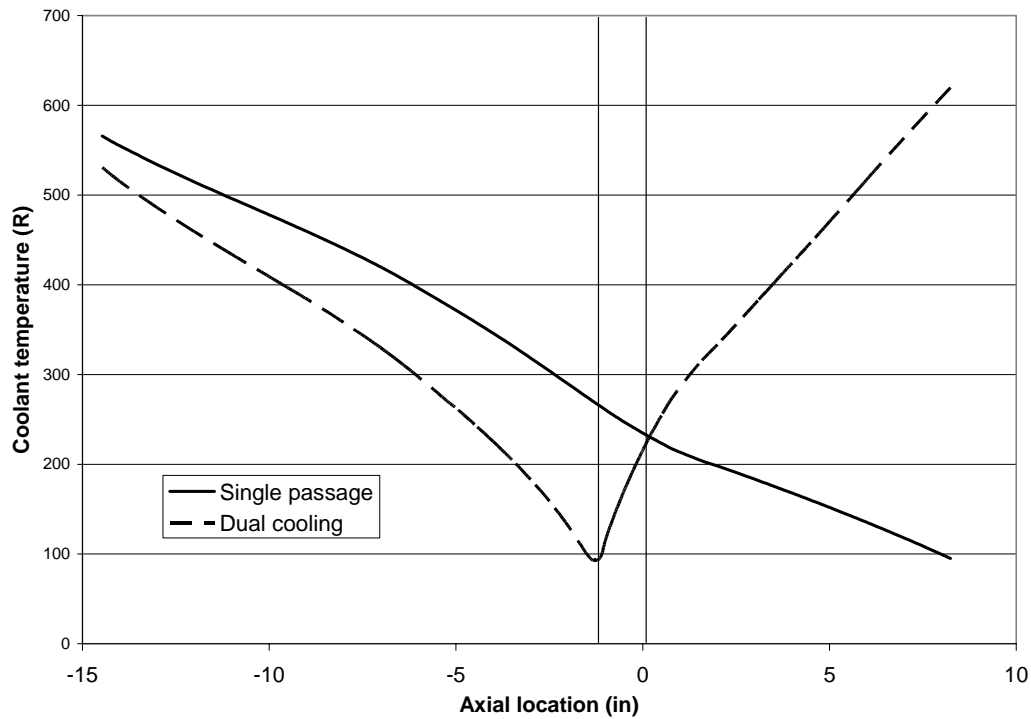
Figure 12. Temperature distribution for single-circuit and dual-circuit designs (dual-circuit cooling channel entrances are placed at the maximum temperature location of a single-circuit system).

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B) Dual-circuit cooling with two different coolants

To demonstrate the effectiveness of dual-circuit cooling using two different coolants, the cooling system of the SSME is retrofitted with separate circuits to accommodate both coolants, LH_2 and LO_2. The redesigned cooling channels have dimensions shown in Figure 13. As shown in this figure, the cooling channel width for both cases (single and dual circuit designs) is the same as the original SSME single-circuit cooling channel width. Both coolants enter upstream of the throat at x = -0.8". For this concept, the liquid hydrogen travels through the upstream cooling circuit and the liquid oxygen travels through the downstream circuit.

The resulting maximum wall temperature distributions for both the original SSME single-circuit and the new dual-circuit designs are shown in Figure 14. From this figure it can be seen that the maximum wall temperature for the new design is reduced by 250 ºR. As shown in Figure 15, this reduction in the wall temperature is due to the lower rises in the coolant temperature with respect to single-circuit design. The temperature rise for hydrogen in the single-circuit design is 471 ºR; while for the dual-circuit channel the temperature rise is only 327 ºR. The temperature rise for the liquid oxygen is 101 ºR.

The variations of coolant pressures for both cooling circuits are shown in Figure 16. The coolant pressure drop for the single-circuit is lower than that of the dual-circuit by 548 psi (2774 psi – 2322 psi). Although the pressure drop is larger for the dual-circuit design, the dual-circuit coolant stagnation enthalpy at the exit is substantially lower than for the single-circuit channel; 1410 BTU/lbm versus 1950 BTU/lbm.
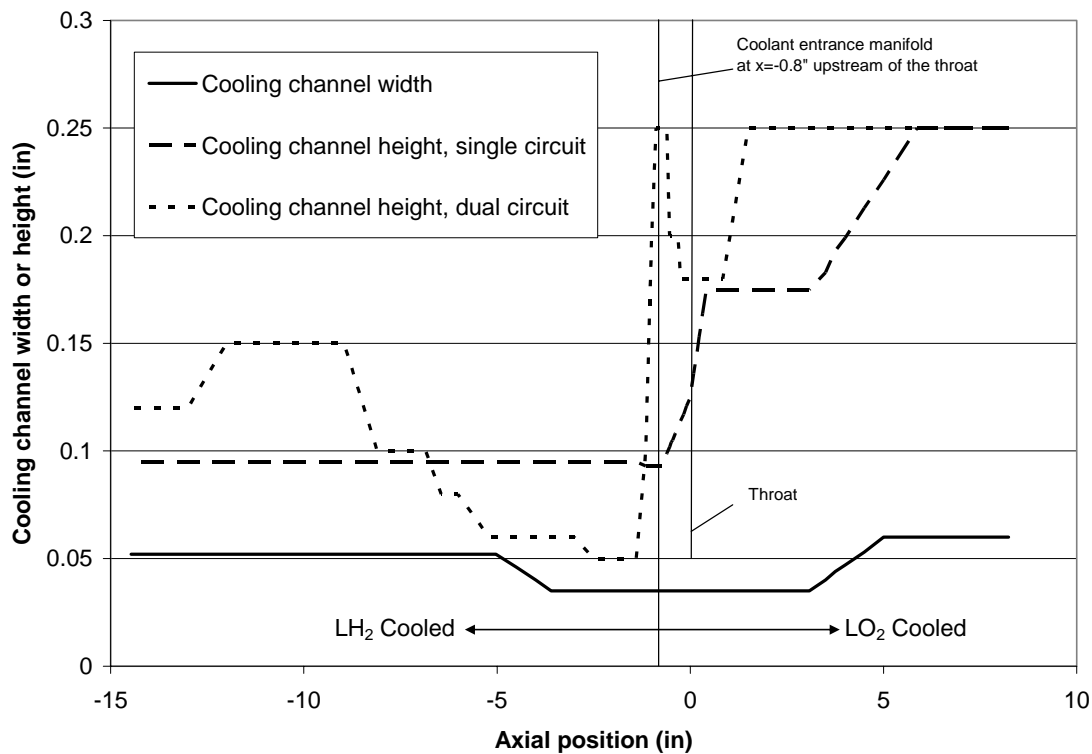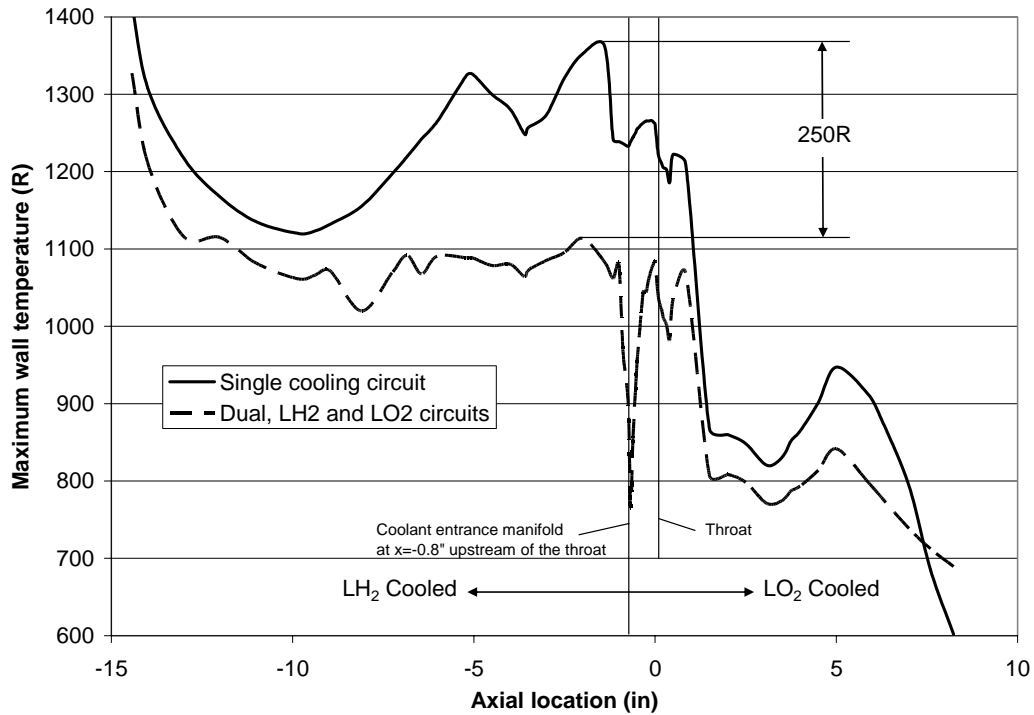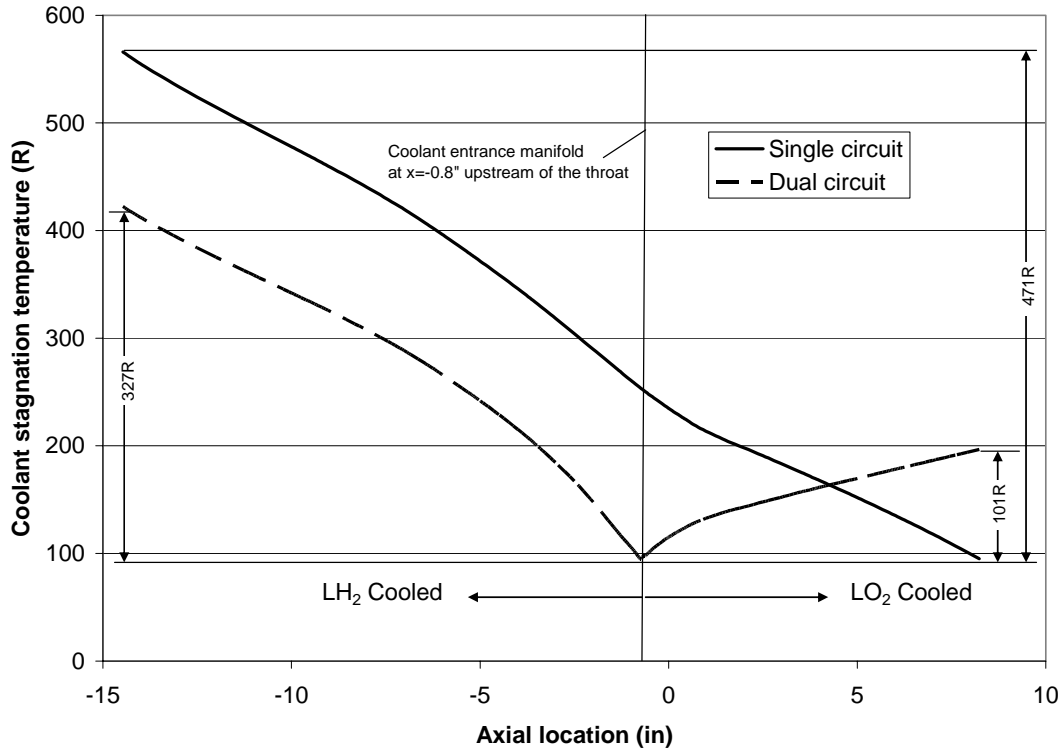

Figure 13. Dimensions of the SSME dual-circuit with two different coolants.

Figure 14. Maximum wall temperature distribution for both the original SSME single-circuit and the new dual-circuit designs; using both LH$_2$ and LO$_2$ as coolants.

Figure 15. Coolant temperature variation along the cooling channels of both the original SSME single-circuit design and dual-circuit designs.
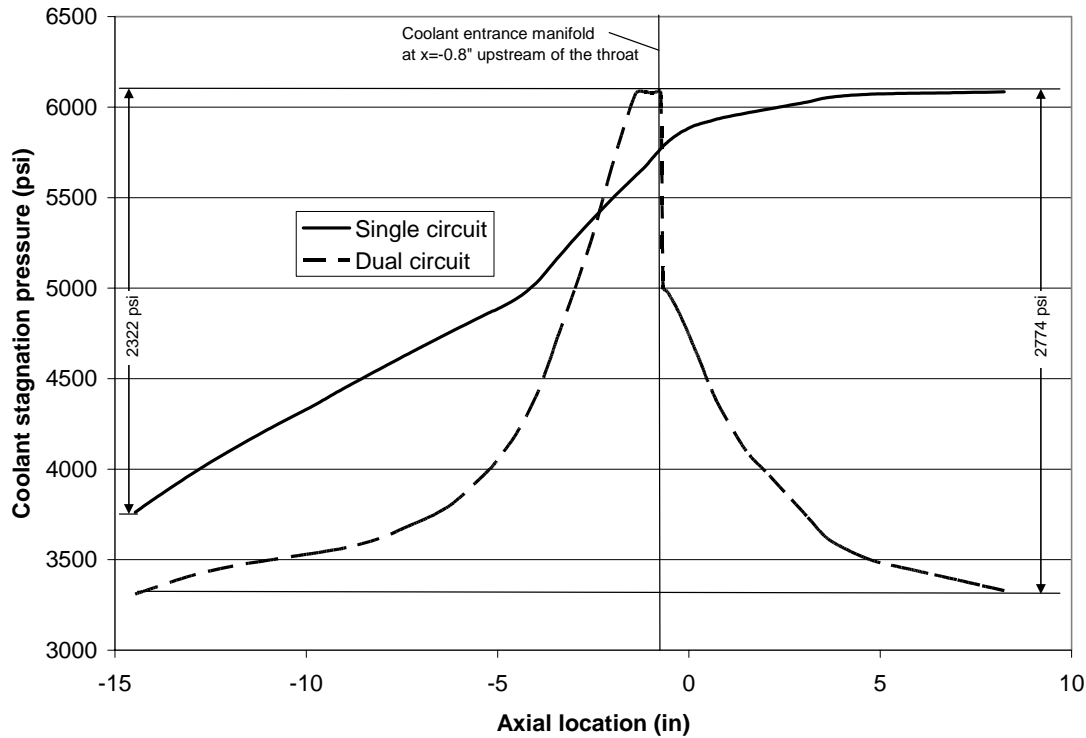
Figure 16. Coolant pressure variation along the cooling channels of both, the original SSME single-circuit and dual-circuit cooling designs.
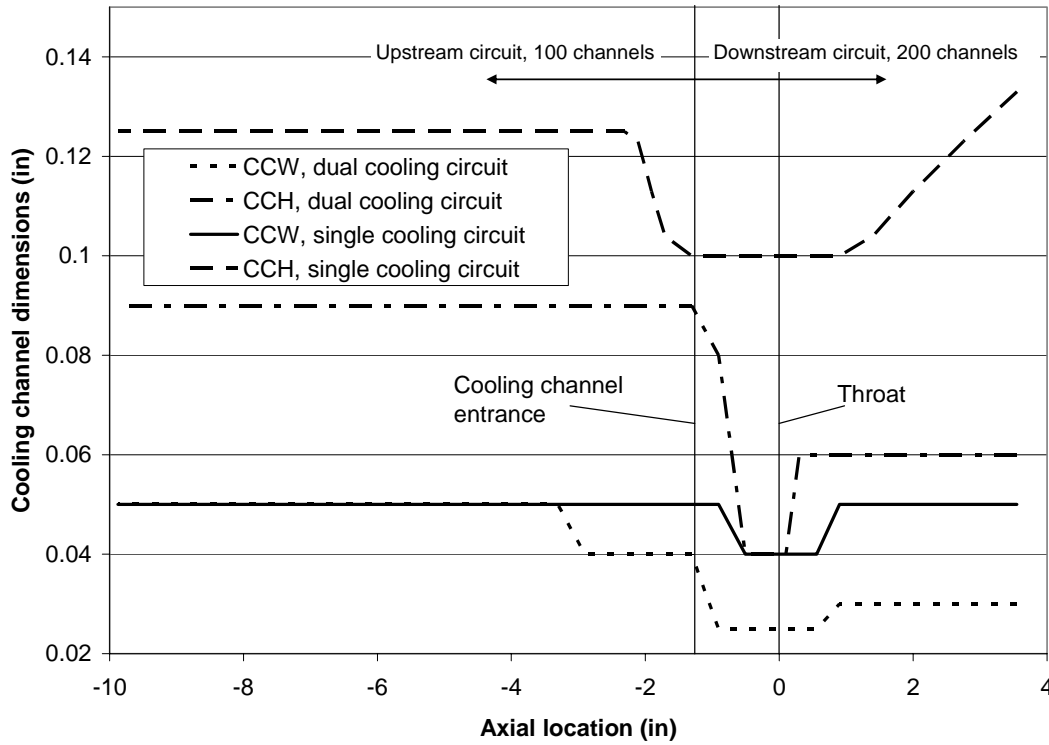
C) Dual-circuit cooling channel RP1 cooling

A dual-circuit design can be more efficient than a single-circuit design when the coolant is a hydrocarbon because: 1) lower wall temperature, 2) reduction of the likelihood of coking, and 3) lower coolant pressure drops. To demonstrate the effectiveness of such a dual-circuit cooling design for a hydrocarbon coolant, consideration is given to the same engine described in reference[5].

The RP1 cooled case is analyzed for a coolant flow rate of 20.3 lbm/s and an inlet temperature of 520 °R. The coolant pressure at the entrance of the cooling channel is 1000 psi. The cooling channel wall temperature for RP1 cooling, which is based on the original single-circuit design, exceeds the coking limit (1360°R) set by the Rocketdyne report. Also, the hot-gas-side maximum wall temperature exceeds the NAROY-Z's limit (1560 °R). To lower the wall temperature, the cooling channel height must be decreased to increase the coolant velocity and subsequently increase the convective heat transfer coefficient. However, this increases the coolant pressure drop, resulting in a higher pumping power requirement. The other option is to include a thin layer of Zirconia (0.002 inch). Although this approach is very effective in lowering wall heat fluxes and temperatures, there is the possibility that the thin coating layer could be eroded by high pressure and high velocity hot gases.

To examine the effectiveness of dual cooling circuits in lowering the wall temperature of the RP1 cooled engine, a dual-circuit for this engine was designed. The dimensions of the cooling channels for both designs, single and dual circuits, are shown in Figure 17. The number of cooling channels for the single-circuit is 100. The number of cooling channels for the upstream circuit of the dual-circuit is 100 and for the downstream circuit is 200. Figure 18 shows the maximum cooling channel wall temperature distribution along the axial direction. As shown in this figure, the dual-circuit design reduces the maximum coolant side wall temperature by 212 °R. As a result, this substantially reduces the likelihood of coking. Figure 18 also shows that the coolant side wall temperature for the single-circuit design exceeds the coking limit by 154°R (an unacceptable design). The coolant wall temperatures for the dual-circuit at all locations are below the coking temperature limit.

The maximum wall temperatures on the hot-gas-side for both designs are shown in Figure 19. This figure shows that the maximum wall temperature for the single-circuit exceeds the material limit (1560 °R). The dual-circuit, however, reduces the maximum wall temperature by 191 °R (72 °R below the material limit). Note that the wall heat fluxes at the throat region, as shown in Figure 20, increase slightly for the dual-circuit due to the lower wall temperature, since the dual-circuit design results in a lower wall temperature than that of single-circuit design.

Figure 17. Cooling channel dimensions of the RP1 cooled engine for the single-circuit and dual-circuit designs.
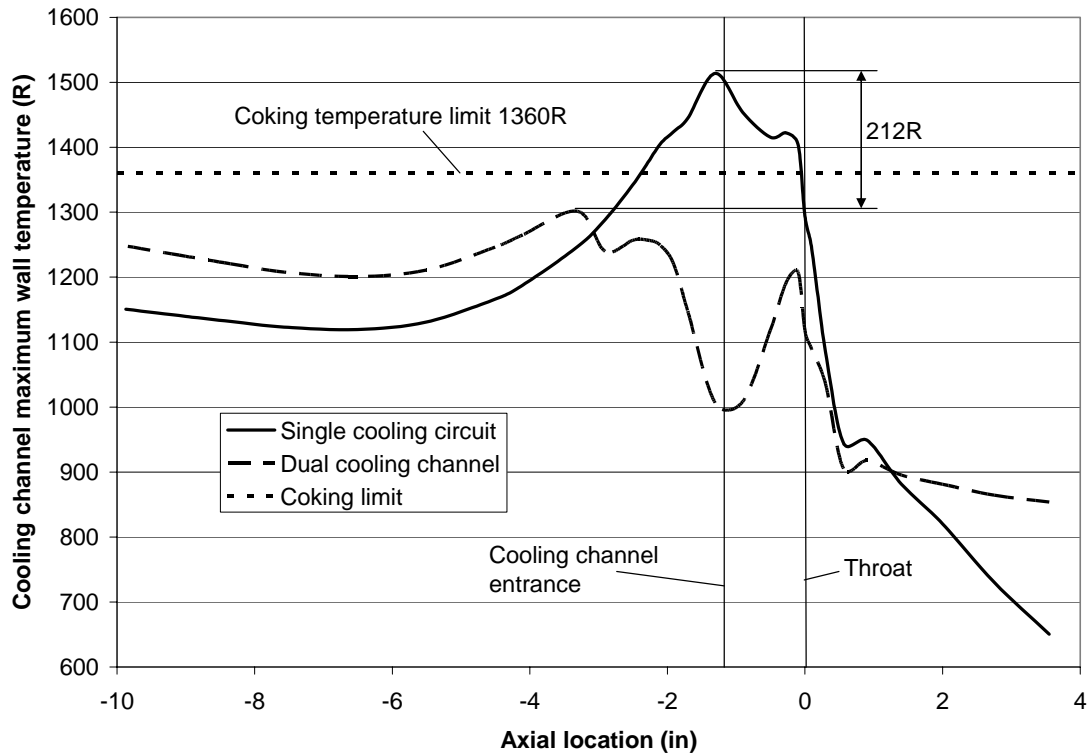
Figure 18. Maximum coolant side wall temperature for both single and dual circuits.
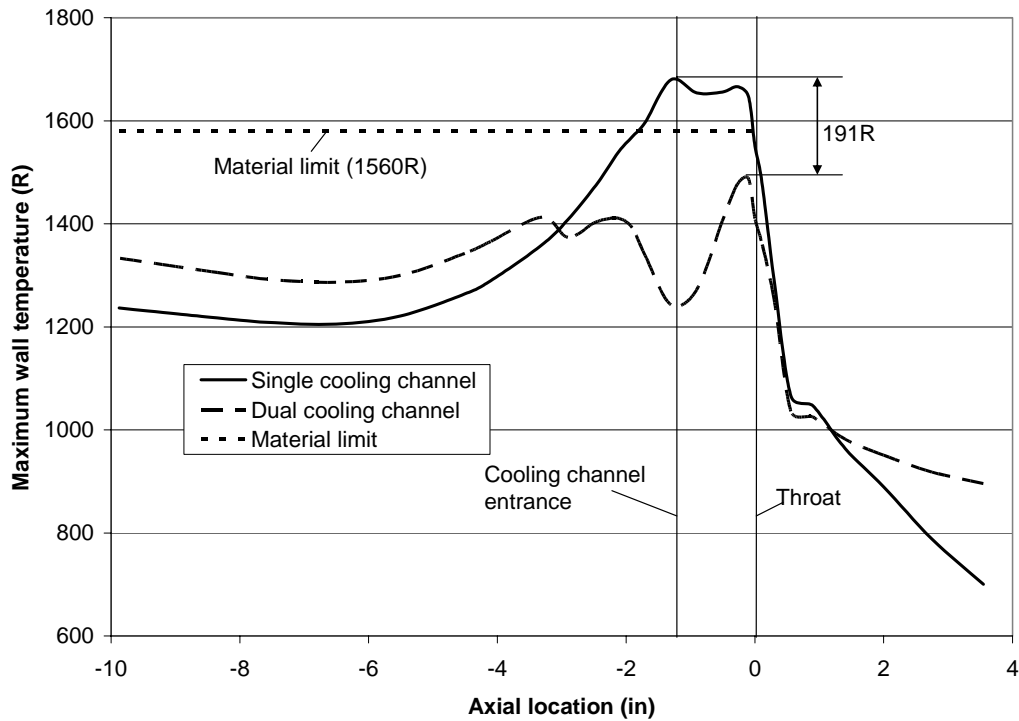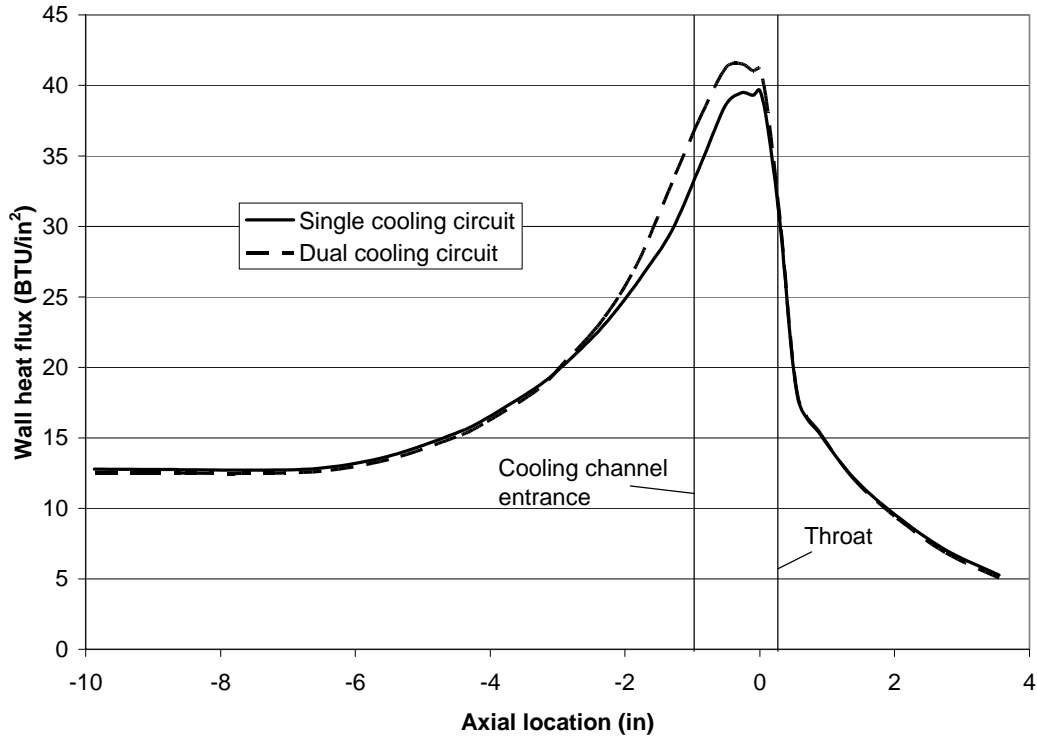

Figure 19. Maximum hot-gas-side wall temperature of the RP1 cooled engine for single and dual circuit designs.
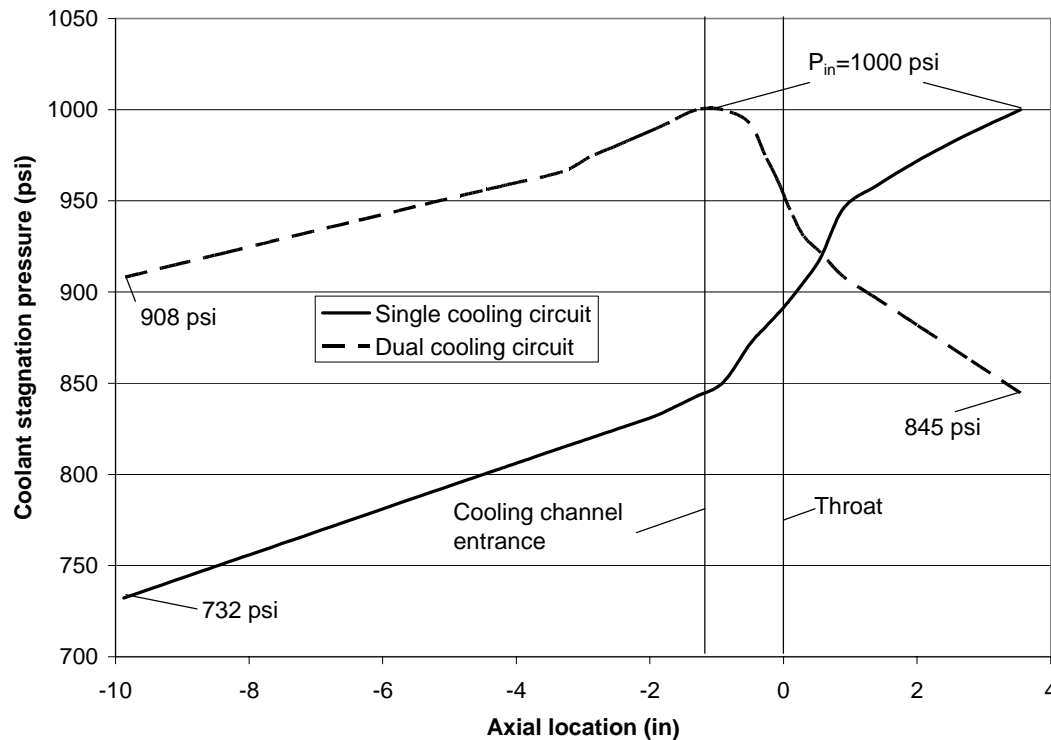
Figure 20. Wall heat flux along the axial location of the RP1 cooled engine for single and dual-circuit cooled engines.

An important aspect of the dual-circuit cooling system is the reduction in wall temperature combined with lower coolant pumping power requirements. Figure 21 shows the stagnation pressure variation along cooling circuits for both single and dual circuit designs. As shown in this figure, the inlet coolant pressure is 1000 psi for both cases. The pressure drop for the single-circuit is substantially higher than for both channels of the dual-circuit cooling design (268 psi for single-circuit versus 92 psi for the upstream circuit and 155 psi for the downstream circuit of the dual-circuit design). The pumping power can be calculated via the following equation:

$$\text{Power} = \frac{\dot{m}\Delta p_0}{\rho} \tag{2}$$

Based on the above equation the pumping power required to push the coolant through cooling channels of the single-circuit design is 31.67 hp. Similarly, the power required for the downstream and upstream circuits of the dual-circuit design are 8.12 hp and 6.07 hp, respectively, resulting in a total power of 14.19 hp for a dual-circuit design. This indicates a 55% reduction in coolant pumping power when using a dual-circuit design.
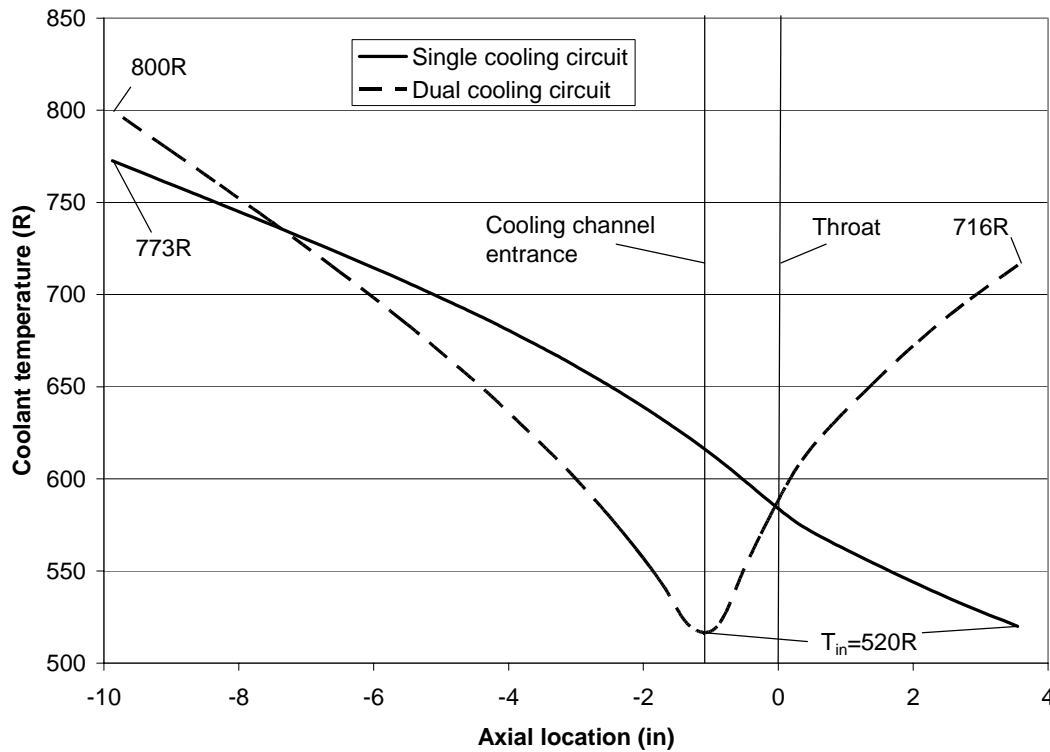
Figure 21. Coolant pressure along the axial location of the RP1 cooled engine for single and dual-circuit cooled engines.

The coolant temperature variations along the axial direction for both designs are shown in Figure 22. The entrance coolant temperature for both designs is 520 ºR. The exit coolant temperature for the single-circuit design is 773 ºR, while it is 800 ºR for the upstream cooling circuit of the dual-circuit design, and for the downstream circuit it is 716 ºR. The resulting coolant mixture temperature for the dual-circuit based on Equation (1) is 760 ºR, while the exit temperature for the single-circuit is 773ºR; resulting in the dual-circuit cooling mixture temperature being 13 ºR less than the single-circuit temperature.

Another important characteristic of RP1 cooled engines is the coolant velocity in the cooling channels. The coolant velocity must be kept higher than 70 ft/s in order to avoid coking. Figure 23 shows the coolant velocities for both dual-circuit and single cooling circuits. The coolant velocity for both designs is more than 70 ft/s.

The dual-circuit cooling design with two coolants, one coolant RP1 and the other one liquid oxygen, is impractical. This is due to the fact that at the entrance of the cooling channel liquid oxygen is at a very low temperature (95 ºR) and RP1 is at room temperature. Having both coolant manifolds next to each other results in the freezing of RP1.

Figure 22. Coolant stagnation temperature along the axial location of the RP1 cooled engine for single and dual-circuit cooled engines.
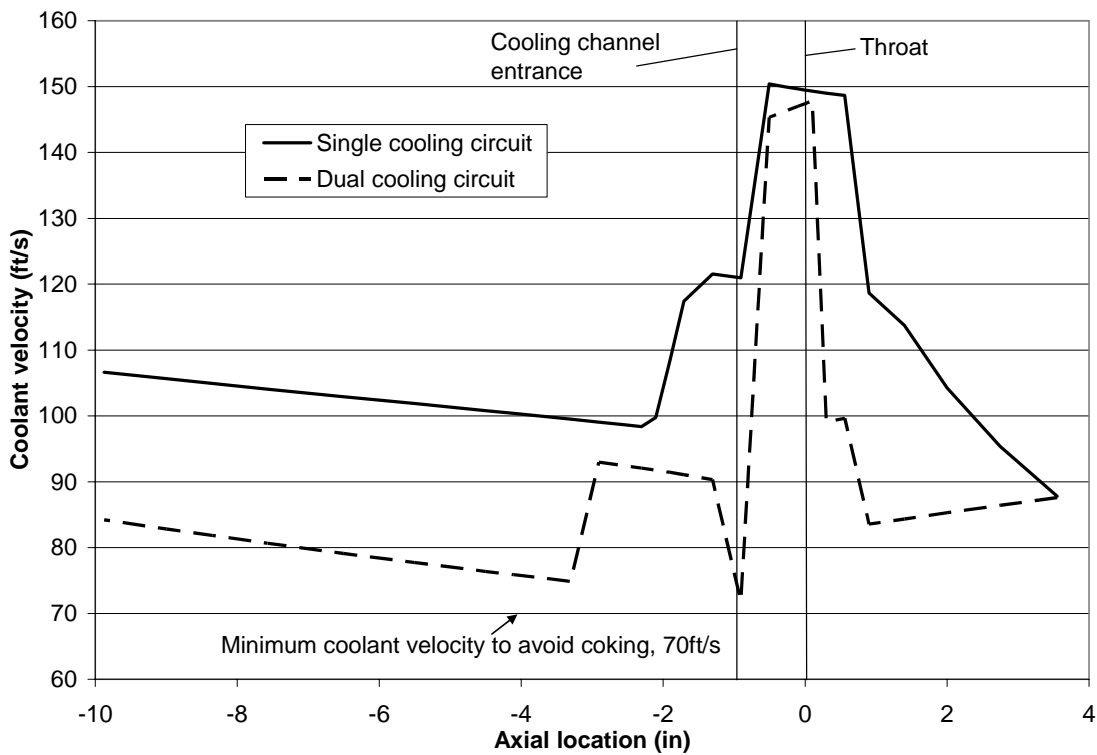
Figure 23. Coolant velocity along the axial location of the RP1 cooled engines for single and dual-circuit cooled engines.

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III. Concluding Remarks

The effectiveness of regenerative dual-circuit cooling designs is studied. The dual-circuit is shown to reduce wall temperatures at the throat area for the SSME and a RP1-LOX engine. The reduction in wall temperature is accomplished by a lower coolant pressure drop, resulting in lower coolant pumping power.

It is also shown that the dual-circuit cooling design reduces the likelihood of coking and substantially reduces the wall temperature when using RP1 as the coolant. The only disadvantage of dual-circuit cooling is the manufacturing cost of an additional manifold for the downstream circuit. In some engines the downstream coolant circuit can be used as dump cooling (see reference 1 for description of dump cooling), hence eliminating the need for an additional manifold.

IV. Acknowledgment

V. References

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